THE UNIVERSITY OF ALBERTA

EFFECTS OF BALLAST ON TRACTION PARAMETERS

by

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The undersigned certify that they have read, and recommend to the Faculty of Graduate Studies and Research, for acceptance, a thesis entitled "Effects of Ballast on Traction Parameters" submitted by Robert Donald Borg in partial fulfilment of the requirements for the degree of Master of Science.

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ABSTRACT

Ballast, speed and surface conditions were studied for their effects on the tractive performance of a standard agricultural tractor on an Alberta soil. The tractor was a Massey Ferguson 135, the ballast range was 930-3840 pounds, the speed range was 1.5-3 miles per hour, and the surface conditions were firm grass and loose fallow.

Performance parameters (coefficient of net traction, tractive efficiency, tractive power coefficient, and drawbar horsepower) were evaluated statistically by an analysis of covariance and by a stepwise multiple regression of the data.

The following observations were made:

1. The coefficient of net traction (from analysis of covariance at 37% slip) decreased with an increase in ballast. Speed had a small effect on the coefficient.
2. The ballast level at which the tire was loaded to its maximum capacity (14 psi pressure) produced the maximum tractive efficiency. Efficiency increased slightly as ballast was increased and decreased as speed was increased.
3. The drawbar horsepower varied directly as the tractor speed and ballast level. Within the ballast and speed range used in this study, the tractor's engine power was not completely used. The speeds and ballast levels chosen for study did not cause high enough rolling resistance losses to decrease the maximum drawbar horsepower.

4. An estimate of ballast required to utilize all the available engine power was proposed.

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1. INTRODUCTION

1.1 Purpose

The major object in traction is to transfer a large horsepower at a low speed to a given surface. The traction characteristics of this surface may vary and may not always be favorable to an efficient transmission of power.

Most farmers add weight or ballast to the tractor. Insufficient ballast may not allow the full potential of the tractor to be realized due to excessive slippage while excessive amounts of ballast may cause a rolling resistance loss robbing a valuable proportion of the available power. At certain levels of ballast a highly efficient transfer of power may be possible yet the work output may remain low. In other cases the converse may be true. The purpose of studying the effects of ballast are to determine for a surface the amount of ballast that gives the best output performance whether it is pull, efficiency or work output subject to acceptable constraints such as maximum wheel slip and engine power.

1.2 Objectives

The objectives of this study were to:

1. determine an optimum level of ballast with respect to efficiency and power output,
2. investigate the effects of changes in tractor speed and surface conditions on traction parameters,
3. endeavor to analyze traction tests by an existing statistical method.

2. LITERATURE REVIEW

The goal of traction research is to be able to design and use traction devices. In working towards this goal, researchers have been trying to develop a complete traction mechanics. The mechanics will be a function both of the traction device and the soil upon which it acts.

2.1 Approaches to Traction Research

Two approaches can be used in the development of prediction equations. One method is to develop the equations from basic forms of behaviour in which soil strength relations represent the soil and geometric relations describe the traction device. The second method is to develop empirical relations based on field observations.

2.1.1 Analytical Approach

A reasonable method for prediction should be to try and relate soil properties to wheel behaviour. The starting point has been the classical Coulomb soil strength relation; Smax = C + P tan ϕ,

where Smax = maximum shear stress in the soil (psi),

C = cohesion (psi),

P = pressure normal to the shear plane(psi),

and, ϕ = angle of internal friction (degrees).

Micklethwaite as cited by Janosi and Hanamoto (1961) multiplied both sides of Coulomb's equation by the contact area to arrive at a thrust prediction equation. This relation predicts maximum thrust based on maximum soil strength but does not consider relative displacements. To overcome this difficulty and to present a more general prediction equation Bekker (1956) developed a relationship describing the shape of a soil stress - displacement curve. The thrust of a wheel is predicted by

assuming a pressure distribution on the soil then integrating the shear stress along the soil - wheel contact surface.

Bekker's equation is involved and includes the case where a soil stress - deformation curve exhibits a hump and a decay. Since most soils do not follow this pattern, Janosi and Hanamoto (1961) have proposed a simpler relation. The relation is:

$$S = (C + P \tan \phi)\ (1 - e^{-j/k})$$

where S = shear strength (psi),

j = soil deformation in the horizontal direction (in.),

k = deformation modulus (in.)

derived from a soil stress - displacement curve. The displacement was expressed in terms of slip, then the shear stress was integrated over the contact length to give thrust as a function of slip.

Although the analytical approach may seem to be the most desirable, there are problems. Leflaive (1966) summarizes the difficulties. Many of the necessary physical laws for a proper analysis are not at hand. Differences in behaviour between tires may be due to differences in the complex soil flow patterns. The actual mechanical properties of a pneumatic tire are not easy to define. Stress - displacement curves for a particular experimental set-up on soils are known but not the stress - strain curves. Soil mechanics at the present time does not consider rapid and extensive soil deformations.

2.1.2 Empirical Approach

While an analytical approach may include all the factors that the designer thinks are important for prediction, it cannot isolate new design factors and may not be sensitive enough to distinguish between design variables. For the sake of illustration a new design factor

was the introduction of radial ply tires while differences between design variables was the example of lug angle changes. The empirical approach by itself has the disadvantage that it is primarily a comparative procedure and may not be able to predict performance (Gill and Vanden Berg, 1961).

Persson (1967) suggests that the behaviour of the wheel should be expressed by parameters independent of the vehicle used yet still describe the influence of the wheel on the vehicle. These parameters can predict wheel performance without directly correlating the performance to soil properties. The philosophy is to first obtain the wheel performance data, then to correlate this data with soil properties.

2.2 Effects of Ballast

Vasey and Naylor (1958), Leflaive (1966) and Persson (1967) have demonstrated that adding to the basic weight of the tractor decreases slippage and increases both drawbar pull and horsepower (when engine power is not limited). The current design philosophy seems to produce tractors with a high power to weight ratio so as to have engine power available for hydraulic and power - take - off (PTO) applications. The tractor is also designed to operate on a variety of soil conditions. Depending on the geographical area of operation (soil texture and geological make-up influences)(Williams and Van Syoc, 1968), different levels of wheel slip are acceptable, various tillage tools may be used, and speeds of operation (weight required varies inversely as the speed) change. With these changing conditions ballast requirements to give optimum performance vary.

An excessive amount of ballast may cause a degree of soil compaction detrimental to plant growth (Feldman and Domier, 1970). Permeability to

water and aeration is reduced while mechanical strength is increased. The amount of compaction is a function of the soil moisture content. Cohron (1971) notes however that there is not a satisfactory theory at present relating soil-wheel behaviour to compaction. Compaction is also a function of soil shear (Gill and Vanden Berg, 1967). Therefore a lack of ballast causing high wheel slip may also be detrimental.

2.2.1 Effects of Soil

Kliefoth (1966) summarizes the effect of soil type on the performance of a tire after weight is added. The traction coefficient (defined by Kliefoth as the driving force divided by the real load on the tire) decreases when load is increased when the tire acts on soils of a poor bearing capacity (moorland). The coefficient increases as load is increased when the tire acts on soils of a good bearing capacity (compact, damp, cohesive soils). On frictional soils (sands) the traction coefficient remains nearly constant as the load changes.

Freitag (1965) found that if deflection was held constant that the effect of changing the load on the tire had little effect while operating on friction soils. On non-frictional soils (very wet mud, muddy, or standing water) the pull to weight ratio decreased with additional weight.

2.2.2 Types of Ballast

Once the decision has been made to add ballast, the next step is to decide on the type of additional weight. One can choose from either dry powder, liquid, or cast iron ballast.

Reed et al (1953) found that if static load and inflation pressures were held constant between tires, all types of ballast had the same performance in terms of traction. Tests were made on concrete, sand,

sandy loam and clay soils.

When operating on concrete, grassland, or fallow surfaces Kucera and Jamison (1965) concluded that differences between the performance of tires ballasted with liquid, powder, or cast iron were not significant.

Another factor to be considered is the effect of ballast types on tractor stability. Beckwith et al (1962) and Shukla (1967) have shown that tractors may be more or less stable depending on how ballast is added. Shukla found that a tractor with 75% liquid filled tires had better roll stability than one with 100% air filled tires, lead ballast tires caused a greater pitch instability than air filled or 75% liquid filled tires, and a lead ballasted tractor was most stable on level ground but not on sloping ground or turns.

2.3 Methods to Find the Optimum Tire Loading

Quantitative methods are necessary for arriving at the best tire load either to give standardized results in traction tests (Kliefoth, 1966) or to help the farmer maximize tractor efficiency.

2.3.1 Analytical Methods

Reece (1967), in finding the best tractive condition, used the theory developed by Bekker and Janosi in which the thrust

$H = (blc + V \tan \phi)X$. X is a slip function

$(X = 1 + \frac{k}{il} e^{-il/k} - \frac{k}{il})$. Drawbar horsepower was expressed as

$$DBHP = \frac{s}{550} (1 - i) [(blc + V \tan \phi)X - R].$$

Variables were:

H = gross thrust (lbs),

b = width of tire contact (in),

l = length of contact (in),

c = cohesion (psi),

V = weight on drive wheels (lbs),

ϕ = angle of internal shear resistance (degrees),

k = shear displacement modulus (in),

i = slip,

s = tractor speed with no slip (ft/sec),

R = rolling resistance (lbs).

Reece took the first derivative of the expression for output power and equated the result to zero. The maximum drawbar horsepower along with the value of slip at this point was found. By incrementing the term V (weight on the drive wheels) one can find either the ballast required for maximum output power or the level of ballast giving the highest power subject to acceptable values of wheel slip.

Harrison (1970) has proposed a model by which one may determine the optimum dynamic weight. The best weight is found when the sum of the power lost to both rolling resistance and slippage is a minimum. The expression proposed was

$$HP = (rm + P_d SM)/375 \qquad (1)$$

where HP = power loss (HP),

P_d = drawbar pull (lbs),

S = slip (%),

r = rolling resistance (lbs),

m = actual speed (mph),

M = no slip speed (mph),

Also $r = C_r W$ where C_r = coefficient of rolling resistance (rear tires)

and W = dynamic rear weight. Harrison proposes a function for slip as $s = k\ (P_d/w)^n$ in which k and n are soil behaviour parameters. By taking the first derivative of equation (1) with respect to dynamic weight the best operating condition may be found. Field trials have not been made to test the model.

The difficulties associated with the analytical approach outlined in section 2.1.1 apply to the preceding approaches. Another difficulty is the lack of an accurate and representative determination of the soil properties over a wide geographical area.

2.3.2 Empirical Approaches

John Deere, in the publication Tractor Field Performance and Ballast Guides (1970) recommends that for the best power output, wheel slip should be held within a 10 - 15% slip range. A procedure is outlined in which charts are used to find the ballast range for full load operation. By trial and error, field measurements of slip along with addition or subtraction of ballasts are made so that the tractor will operate in the best slip range. Although this range is recommended to be 10 - 15% for normal soils, John Deere suggests that 15 - 20% slip may be a better compromise in cases where the tractor is used for lighter loads.

King (1948) presented a series of charts to estimate the performance of a tractor equipped with 10 - 28 rear tires. King used data from tests made by the Allis-Chalmers Manufacturing Company and published by O'Harrow (1947). An assumption was made that the total thrust available from a 10 - 28 tire was directly proportional to the dynamic weight on the rear wheels. With this assumption thrusts at

other loads than those given in the tests could be obtained. King considered the efficiency of the overall system, starting with the fuel burned and ending with a useful drawbar pull. The series of charts consider rolling resistance, thrust available for various combinations of slip and wheel loading, total thrust required for various combinations of rolling resistance and drawbar pull, and traction efficiencies. A chart is given indicating the limitations imposed by the engine.

King found that the maximum tractive efficiency does not vary much through a wide range of loading on clay stubble. Increased weight caused the best efficiency to occur at greater values of pull. On soft surfaces there was a great improvement of efficiency when weight was added. The limitations of the procedure are that a wide variety of soil conditions are not covered and the charts refer to a particular tractor (Ferguson TO-20).

Shields (1952) developed a chart which can be used to determine the weight required on rear tractor tires. The chart is a graphical solution of the physical equations describing the soil-tractor system. To find the rear weight the relation - Load on two rear tires = 375 (HP) (Efficiency) ÷ (MPH)(Coefficient of traction) is used where HP = horsepower of the engine and MPH = desired operating speed. Efficiency is the ratio between drawbar pull and gross thrust while the coefficient of traction is the ratio of drawbar pull to tire load. In constructing the chart, efficiency factors and traction coefficients are given for clay, loam, sandy loam, and dry sand soils. The coefficients are average however and may be better suited to comparative procedures. Another limitation of the chart is that it is not dimensionless and performance of larger tractors cannot be predicted.

Zoz (1972) has also presented a graphical technique to predict performance. Tire efficiency (the ratio of power output to power input of the wheel) and dynamic ratio (the ratio of drawbar pull to dynamic tire weight) along with tractor dimensions may then be used to predict the performance of any tractor on various soil surfaces. Ballast may be determined for desired operating conditions. Implicit in the graphical solution is the assumption that tire tests on one size of tractor may be used to predict the performance of larger tractors. According to Zoz the ballast obtained is still only an approximation.

Friesen and Domier (1968) found that on Manitoba soils the best tractive efficiencies occurred at 16% slip for two wheel drive tractors. The authors recommended that the traction coefficient curves and the tractor characteristics be used to calculate the amount of ballast required on the rear tires. Formulas that were developed based on traction tests for single rear wheels were:

$$\text{optimum rear weight} = \frac{.56 \times (\text{PTO})\text{HP} \times 375}{.40 \times \text{speed (mph)}}$$

on cohesive soils, and

$$\text{optimum rear weight} = \frac{.67 \times (\text{PTO})\text{HP} \times 375}{.425 \times \text{speed (mph)}}$$

on non-cohesive soils. (PTO)HP refers to the engine power delivered at the power-take off and speed is the advertised forward velocity (at rated engine speed) minus 16% for slippage.

The approaches of Zoz, Friesen and Domier estimate ballast requirements based on field tests. The procedures provide a first estimate. The second step is to check wheel-slip under field conditions then add or subtract ballast.

The difficulties associated with the analytical approaches seem to preclude their immediate use. To study the traction conditions of Alberta soils, an empirical approach will be of more benefit. If one wishes to characterize these soils, the influences of ballast, speed and surface condition are important.

3. TRACTION PARAMETERS

The parameters used in this study are the coefficient of net traction, tractive efficiency, tractive power coefficient, and slip. The drawbar horsepower is also considered.

Slip (s) as defined by Gill and Vanden Berg (1967) is the ratio of decreased velocity to the initial velocity (v_o). (v) is the velocity of the vehicle. Then $s = (v_o - v)/v_o$. In this study slip was determined by the ratio of the difference in rear and fifth wheel (actual forward speed indicator) speeds ($v_R - v_5$) to the rear wheel speed (v_R). Then $s = (v_R - v_5)/v_R$. Zero slip was defined as the condition in which the vehicle developed no torque at the rear wheel. The tractor was towed to calibrate the speed transducers. The speeds of the rear and fifth wheels were expressed in miles per hour.

The coefficient of net traction (μ) is defined by the A.S.A.E. (1970) as the ratio of the net drawbar pull (P) to the supporting force of the rear wheel (V). $\mu = P/V$.

Tractive efficiency (η) is the ratio of output power of the wheel (drawbar horsepower) to the input power of the wheel (axle horsepower). Axle horsepower is proportional to the product of the torque (T) and the angular velocity (ω) of the wheel. Drawbar horsepower (DBHP) is proportional to the product of net pull (P) and forward velocity (v_5) of the wheel. $\eta = Pv_5/T\omega$.

Drawbar horsepower (DBHP) = Pull (P) x $v_5/375$ when pull is expressed in pounds and forward velocity is in miles per hour.

The tractive power coefficient (π) can be derived from the other parameters. The tractive power coefficient is defined (Persson, 1967) as $\pi = Pv_5\omega_o/Vv_o\omega$. $\pi = \mu(1 - s)$. This measure (Gill and Vanden Berg, 1967)

includes the influence of both pull and speed and represents the work done. Persson (1967) states that π may be indicative of the output power of the wheel.

4. EQUIPMENT

4.1 Variables

Quantities describing the wheel which were measured in this study:

a) Output

i) net pull,

ii) supporting force

iii) forward velocity.

b) Input

i) torque,

ii) angular velocity.

4.2 Equipment and Instrumentation

The instrumentation and equipment used in this study consisted of:

1. Test tractor: A Massey-Ferguson 135 diesel tractor was used. Tires were 14.9-24 on the rear and 6.00-16 on the front. Complete specifications are given in Appendix 1.
2. Drawbar loading unit: A load car shown in figure 2, built by the Department of Agricultural Engineering,was used to provide a variable load. The car consists of rear axle and transmission assemblies of two tractors. Two meter-out valves control the torque of constant displacement hydraulic pumps which are driven directly by the front and rear wheels of the load car. By varying the flow from the pumps and by choosing an appropriate gear, a desired drawbar load could be maintained.
3. Pull transducer: A steel ring (dimensions in Appendix 1) fitted with four strain gauges arranged in a Wheatstone Bridge

Figure 1: Massey Ferguson 135 tractor.

Figure 2: Tractor, load car, and instrument van.

transferred pull from the load car to the tractor. All strain gauges used in this study were 120 ohm foil gauges. Bridge excitation was 5 volts.

4. Weight transfer: Four strain gauges were placed near the center of the tractors front axle. The gauges were arranged in a Wheatstone Bridge, two gauges in tension and two in compression, treating the front axle as a simple beam.

   The drawbar pull was kept horizontal thus the sum of the static rear weight and the weight transferred indicated the supporting force of the drive wheels.

5. Torque transducer: A transducer (figure 3) built by Berlage (1962) was installed in one rear wheel.

   Torque was transferred from the axle to the wheel through cams consisting of steel balls. The tendency of the balls to ride up in their seats upon the application of a torque is resisted by a cylinder which cannot move laterally with respect to the axle. The compression of the strain-gauged cylinder is proportional to the torque taken by the wheel.

6. Fifth wheel: A direct current generator (specifications in Appendix 1) was driven by a bicycle wheel (figure 4). The voltage generated indicated the forward speed of the tractor.

7. Angular velocity: A direct current generator driven by one rear wheel (figure 3) indicated the angular velocity of the drive wheels.

8. Recording equipment:

   a) An ultra violet recorder, type SE 2005 built by SE

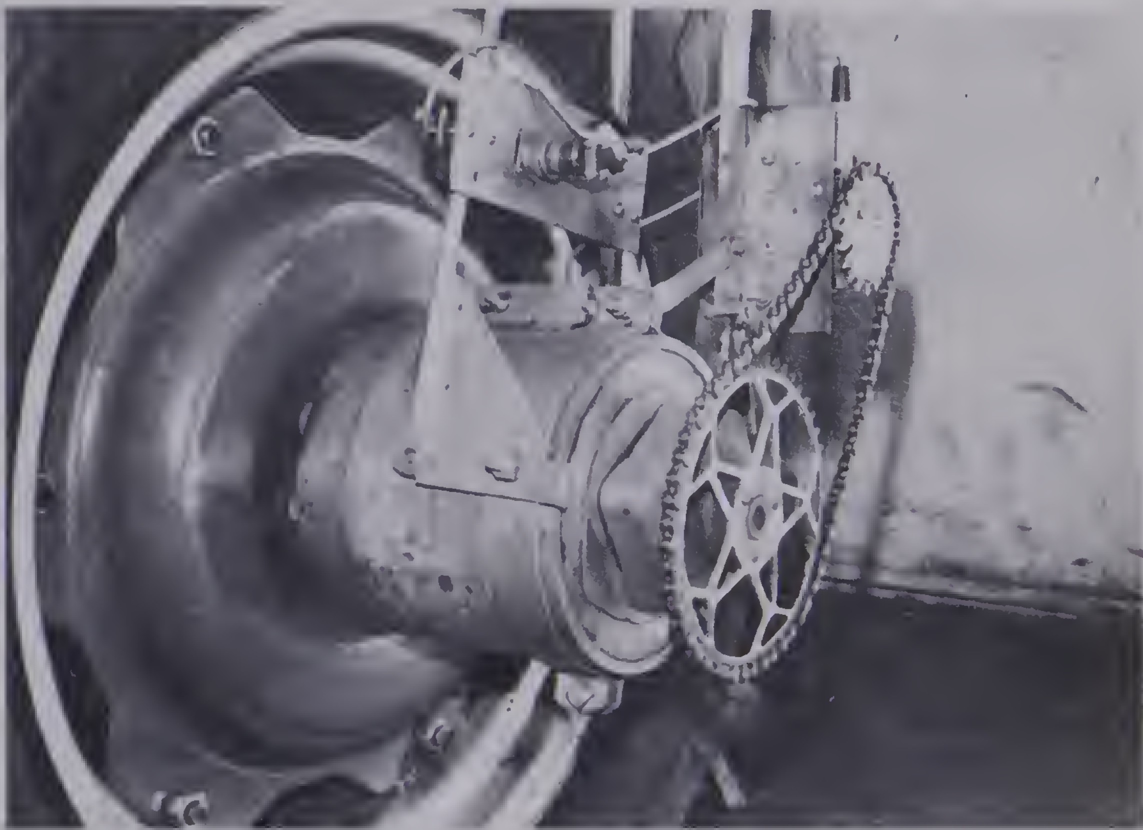

Figure 3: Torque transducer and rear wheel speed indicator.

Figure 4: Tractor with the fifth wheel forward speed indicator.

laboratories, was used to record the electrical signals. Kodak Linagraph Direct Print paper type 1895 was used.

b) Accudata 104DC amplifiers (built by Honeywell) and Accudata 105 Gage control units were used to condition the signals from the Wheatstone Bridges.

9. Power Source: A Kohler electric plant, Model 1.25 mm 25, Series 326386, 1800 rpm, 1250 watts, 120 volts at 60 cycles per second.

4.3 Calibration of Equipment

1. Pull transducer: Calibration was made with a Baldwin universal testing machine. Lines of chart deflection (read from the paper record of the ultra violet recorder) plotted against pull, as shown in figure 5, produced a linear relationship.

2. Weight transfer: Two platform scales were placed under the front wheels. Weight was transferred from the front axle by a hydraulic jack placed under the tractor. Lines of chart deflection were plotted against weight transfer as shown in figure 6. A linear relationship was obtained.

3. Torque transducer: The transducer was calibrated statically. The lever arm shown in figure 7 was attached to the wheel. With the wheel raised off the floor and with the axle locked, weights were added to the arm. Lines of chart deflection were plotted against foot-pounds of torque as shown in figure 9. The shape of the curve was consistent with a calibration curve given by Domier (1966) for the same transducer. Torque was applied to the wheel in the same direction that the torque would be applied in the field.

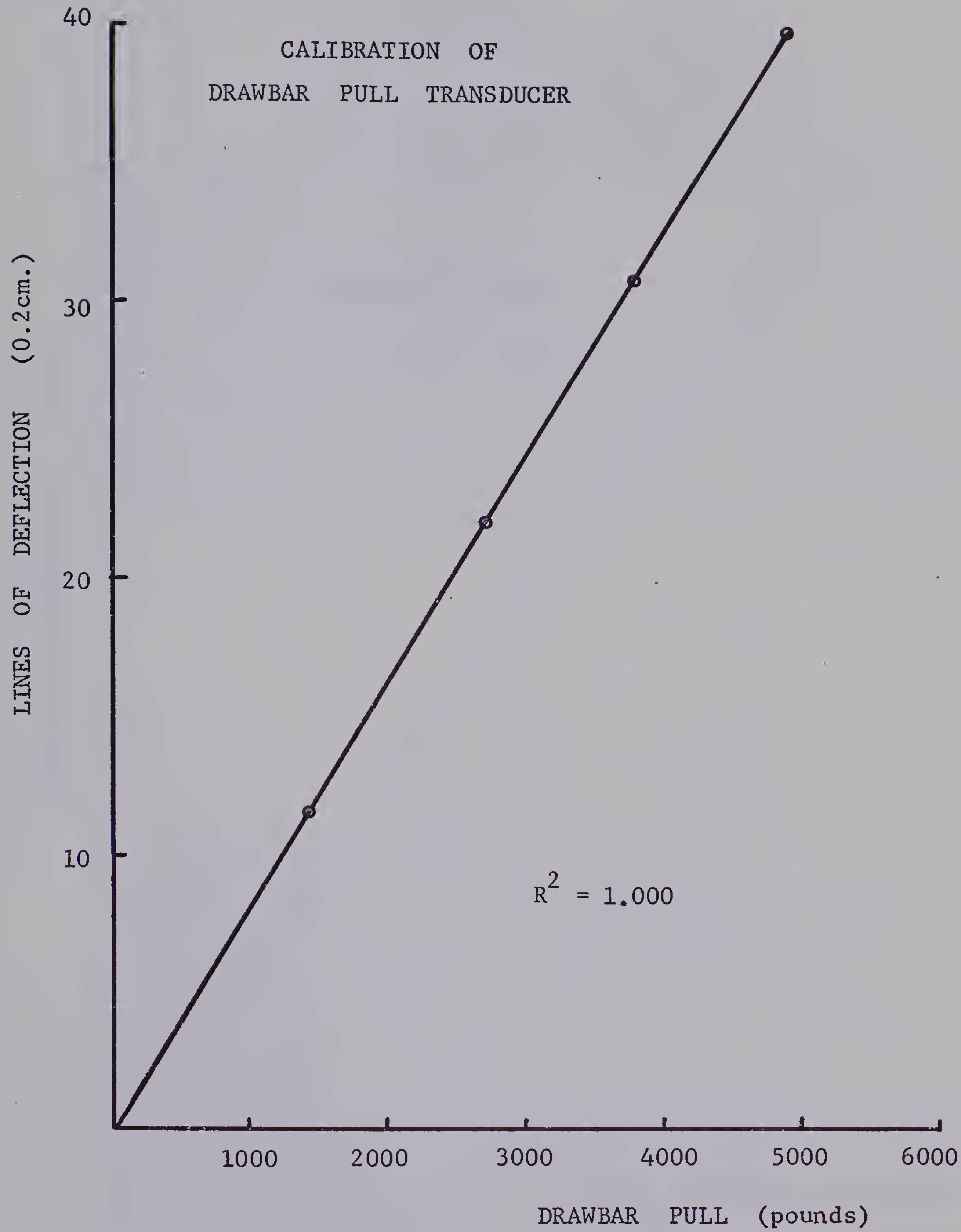

Figure 5: Calibration of drawbar pull transducer.

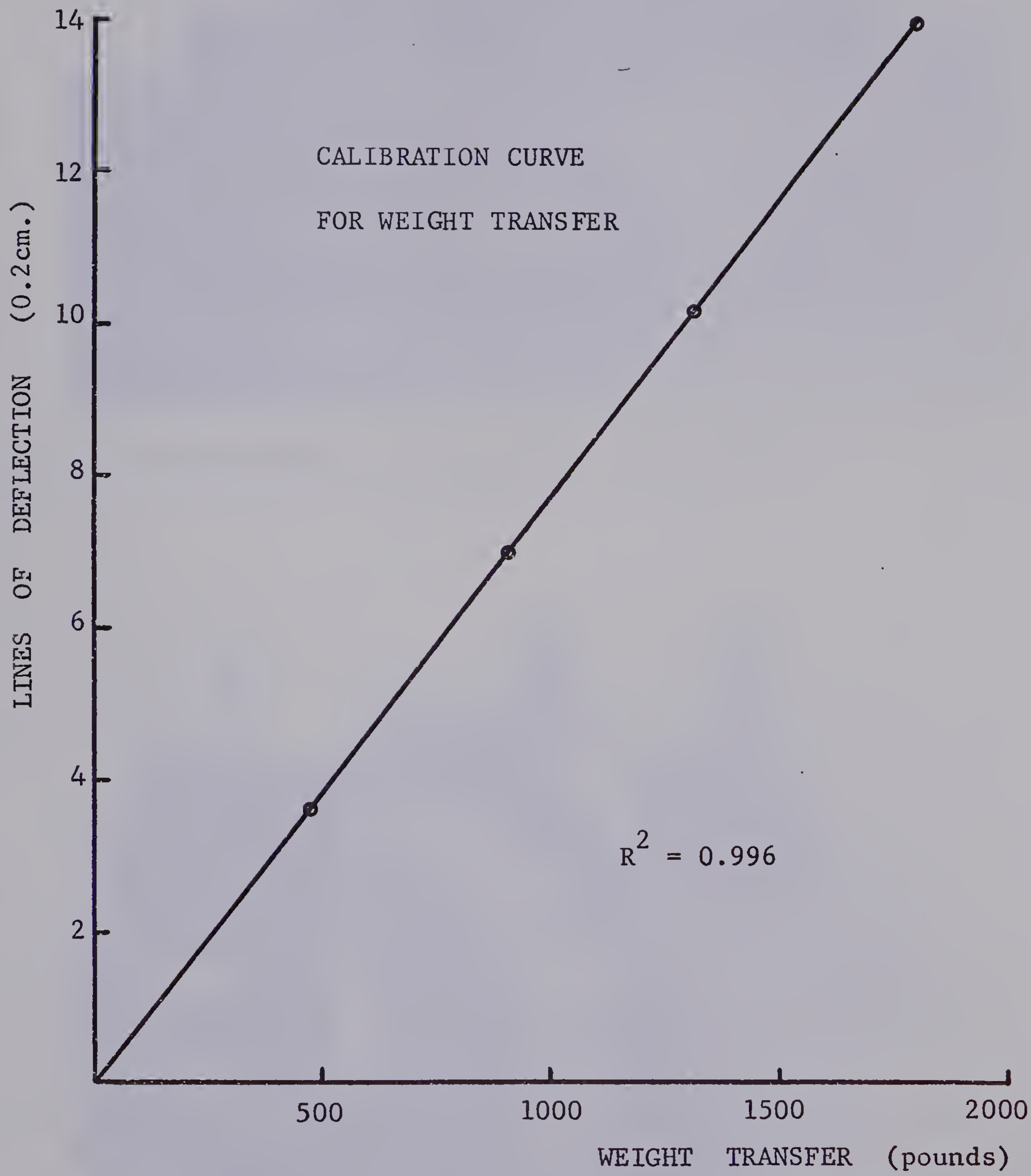

Figure 6: Calibration curve for weight transfer.

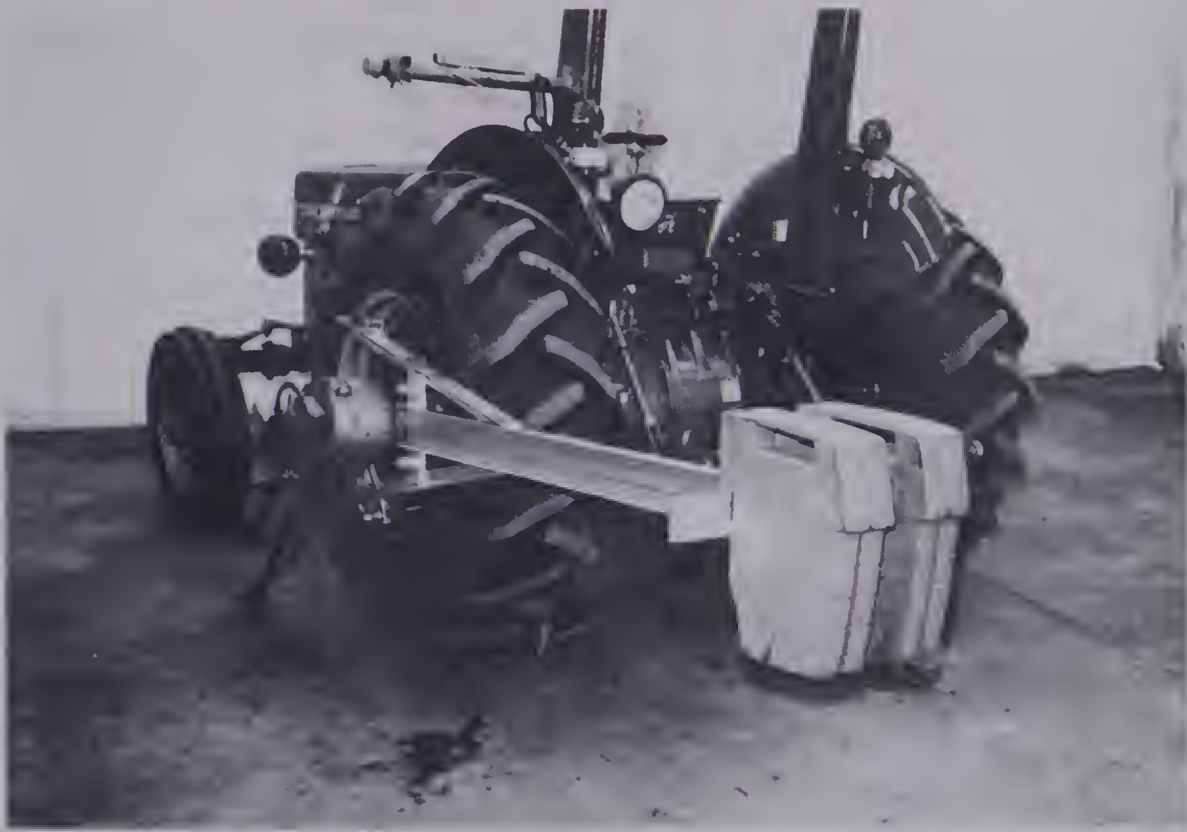

Figure 7: Calibration of the torque transducer.

Figure 8: Tractor with the frame for carrying ballast.

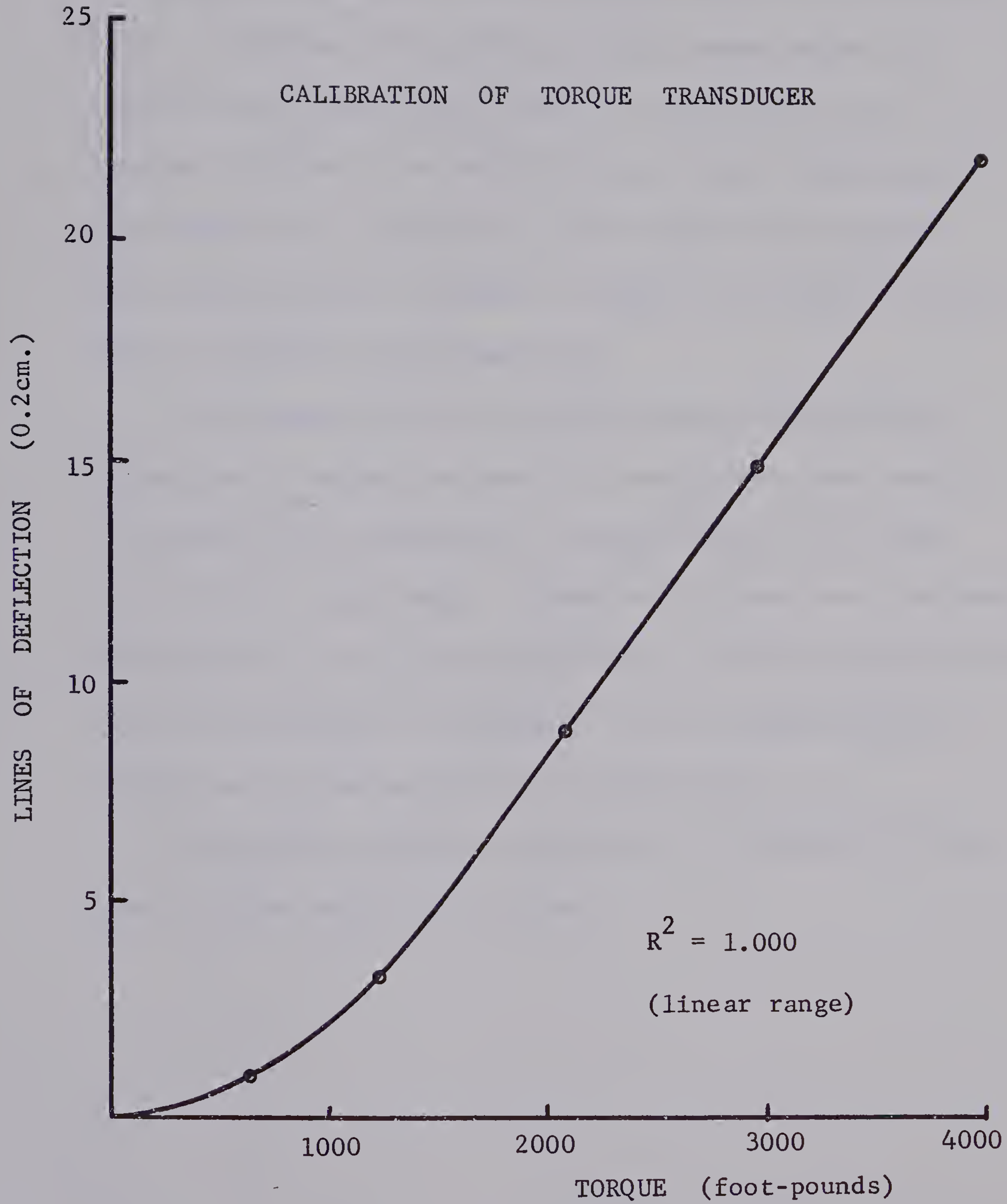

Figure 9: Calibration of torque transducer.

4. Fifth wheel and rear axle speed indicators: Calibration of the rear angular velocity and the fifth wheel forward speed indicators were made on the surfaces on which the tests took place. Ballasts of 930, 2730 and 3840 pounds above the tractors basic weight were used for calibration runs. No apparent difference was noted between curves for the first two weights but a change was noted for the third weight. Tire specifications recommend a higher tire pressure for the 3840 lb. ballast (See Appendix 2).

To determine the relationship between lines of chart deflection and miles per hour the tractor was towed over a one hundred foot distance at a constant speed. The time was noted by a stop watch. Speeds were chosen over the range anticipated in the traction tests and a linear regression was used on the results to determine the calibration curves. A typical set of curves is shown in digure 10.

The squared multiple correlation coefficients (R^2) are given for the calibration curves.

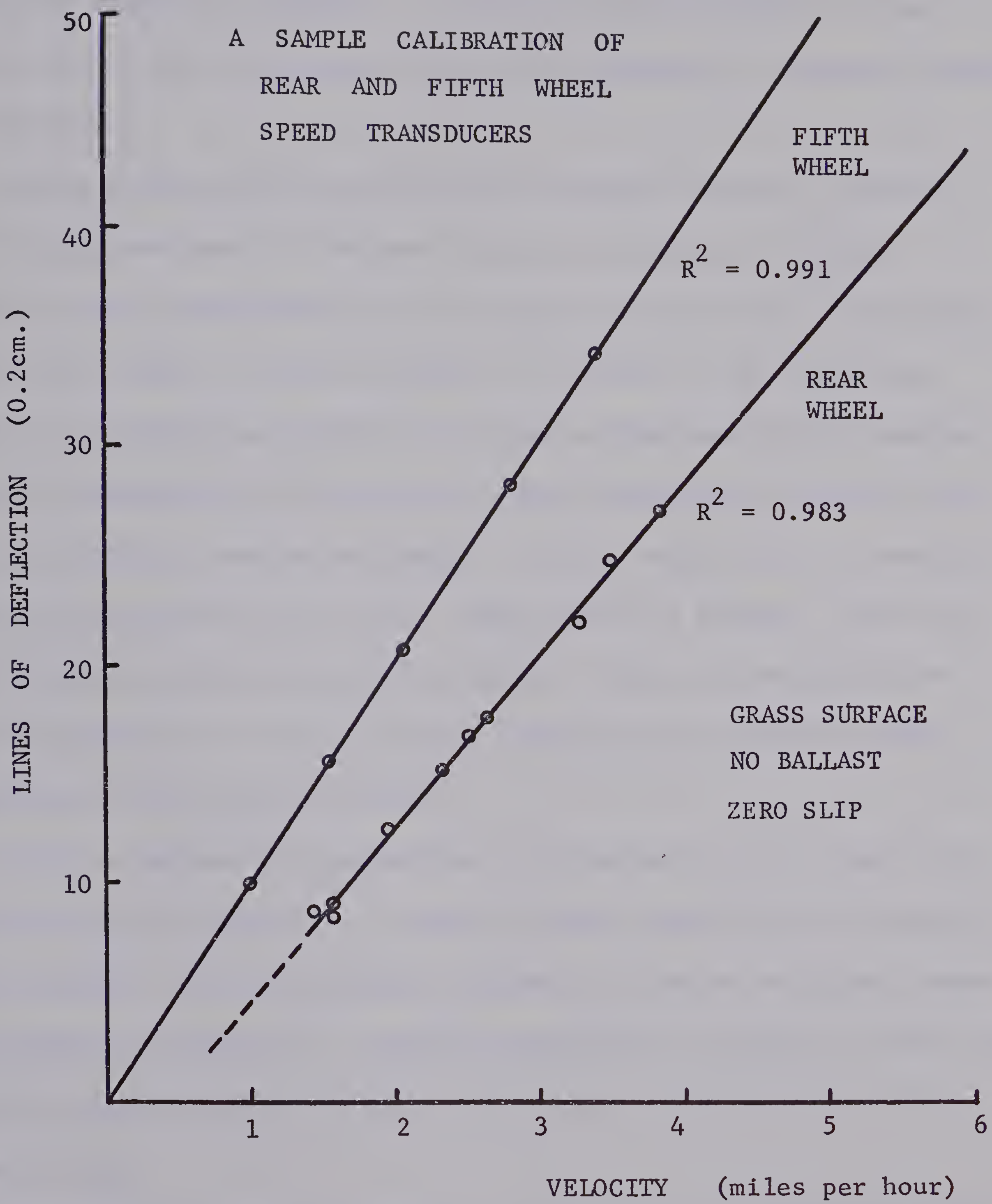

Figure 10: Calibration of speed transducers.

5. EXPERIMENTAL PROCEDURE

5.1 Experimental Design

The experiment was designed primarily to study the effects of ballast on traction parameters. At the same time the effect of two surface conditions of the same soil and the influence of forward velocity was considered.

Liquid ballast (930 lbs. of calcium chloride solution - 75% of tire filled) was added to the rear tires as the range of optimum performance was anticipated to be at higher ballast levels. Cast iron weights were added in nine increments of 364 lbs. to the rear axle. Three 94 lb. weights were added to a frame on the rear of the tractor (figure 8) and one 82 lb. weight was added to the front such that the front axle weight remained unchanged. In this study 1,2,...9 levels of ballast are additions of 0, 364,...,2900 pounds of ballast. The soil was a Malmo clay with a silty clay texture. The soil properties are given in Appendix 3. The two surface conditions were a firm grassy surface and a loose fallow surface.

Within a ballast (B) and surface (S) combination, four gears (G) were chosen (in a range 1.5 - 3 mph) and three repeats (R) were made in each condition. The difficulties involved in a choice of higher speeds are outlined in section 7.6. Twenty observations were made for each run covering a range from low to high slip values. No exact control was placed on slip.

The experiment may be classified as a split plot factorial design. The main effects were surface and ballast. Within a surface and ballast combination the repeats and choices of gears were made.

5.2 Field Test Procedure

The tractor was serviced, the load car was connected, the generator was started and the electronic equipment was allowed to warm up.

Tire pressure was adjusted to recommend levels given by the Goodyear Tire Company (1970) for the ballast level being used.

The prescribed ballast was loaded on the tractor. The tractor was positioned on one of the surfaces, a gear was chosen and a run was made over approximately a 100 ft. distance. The load was continuously varied so that the tractor operated from a condition of low to high wheel slip.

The tractor was run through the series of gears, repeat runs were made and the tractor was moved to the next surface. After both surfaces were completed another increment of ballast was added and the procedure was repeated.

Deflections for torque, pull, weight transfer, rear and fifth wheel speeds were read from the recorder chart for twenty points which covered the entire range of slip used in the test. The results were punched on cards to facilitate computer analysis.

The soil properties given in this study were determined by Subhash Mehra in the University of Alberta Master of Science thesis; 'Traction Characteristics of an Alberta Soil'.

6. ANALYSIS OF DATA

6.1 Evaluation of Performance

To evaluate the performance of the tire with changes of supporting force, forward velocity and soil condition, the performance curves for the various conditions must be compared. In this study the curves for μ, π, η and DBHP were considered. The independent variable chosen was slip. Experimental and historical background influenced this choice.

When evaluating performance it may be desirable to compare the entire performance curves and not one point on the curve. This is sometimes very difficult. A valuable procedure may be the reduction of a performance curve to a single value for use in comparisons. Vanden Berg (1961) suggested some measures such as the maximum coefficient of traction that occurred in a 0 - 70% slip range and the average coefficient of traction for a 0 - 30% slip range. A 0 - 30% range was suggested because most sharp changes in the pull-slip curves occur before 30% slip.

The tractive power coefficient (π), indicative of work output (Gill and Vanden Berg, 1967), may be useful when considered with respect to tractive efficiency. The possibility exists that a tractor may develop a high work output at a low efficiency and conversely develop a low work output at a high efficiency. The desirable point of operation may be where a high work output coincides with a high tractive efficiency.

In order to interpret differences between experimental treatments, statistical methods have been accepted in biological and agricultural studies. Less recognition has been given to this tool in traction studies.

Vasey and Naylor (1958) experimentally obtained slip versus pull curves for five tires on a given surface. Curves were plotted for the

data. The pull was determined for three values of slip (10,25,30% on stubble and 15,25,35% on plowed land). The authors did not desire to compare the tires at a specific level of slip but over the three levels. The means of the three slip values were chosen to test for differences between tires. The means were treated as independent observations and an analysis of variance was carried out. The analysis of variance did not consider the variation of data used in arriving at the means. As a second test the variation of several plotted points from a curve was used to estimate an error variance. The authors discuss the methods and the possibilities of pooled tests using the variances found by both methods. From the final estimate of error variance the t-test is applied to the mean pulls of the tires.

Taylor (1971) used multiple regression techniques described by Birtwistle (1971) to evaluate tractor tire performance. An illustration of a model used was

$$F_x = b_o + b_i L^i + b_j S^j + b_{ij} L^i S^j$$

where F_x is force, L is load, and S is distance per revolution. With the example of load at 3 levels and distance per revolution at 8 levels where i = 1,2. and j = 1,2,3. the model uses 11 predictive variables. During the stepwise regression analysis, reference was made to an analysis of data adjusted within each run to standard values of distance per revolution so that wrong decisions about rejection of non-significant variables could be avoided.

The final stage in presenting the results was a plotting of the coefficient of traction and tractive efficiency curves against wheel-slip. Taylor found the overall power efficiency reached a maximum at

less than 10% slip but since power output is small here, the important region of operation was considered to be 10 - 30% slip.

This study uses the techniques of analysis of covariance and stepwise multiple regression to evaluate the data. The covariance does a regression analysis of the data to compare between treatment effects. The stepwise multiple regression was carried out to determine performance in a 10 - 30% slip range.

6.2 Analysis of Covariance

To evaluate the effects of the variables outlined in the experimental design (surface, ballast, forward speed) on the performance parameters (μ,η,π,DBHP), an analysis of covariance was used.

The analysis reduces the parameters to a common value of the independent variable slip based on a regression analysis of the data. To fit the data the independent variables were slip and the transformation $(\text{slip})^2$. No direct control could be placed on the value of slip (overall mean) to which the parameters were adjusted. The proper choice of data will result in the mean being in the range desired by a researcher.

The model used was:

$$Y_{ijklm} = \mu + B_i + S_j + BS_{ij} + R_{k(ij)} + G_l + BG_{il}$$

$$+ SG_{jl} + BSG_{ijl} + RG_{lk(ij)} + b_1 x_{1ijklm}$$

$$+ b_2 x_{2ijklm} + e_{m(ijkl)}$$ where Yijklm is the m^{th} observation on the i^{th} level of ballast (B) on the j^{th} soil surface (S) on the k^{th} replicate (R) on the l^{th} gear (G).

$i = 1,2,\ldots,9.$

$j = 1,2.$

$k = 1,2,3.$

$l = 1,2,3,4.$

$m = 1,2,\ldots,20.$

x_1 and x_2 are the independent variables slip and slip^2 (transformed variable) respectively. b_1 and b_2 are the regression coefficients.

Standard tests of significance were carried out after the appropriate error terms and mean squares for the split plot design were found (Winer, 1962). A computer program (analysis of covariance for factorial design) (Dixon, 1968) was used to compute the sums of squares of the data.

6.3 Regression Analysis

To arrive at curves of the various parameters versus slip a stepwise multiple regression was carried out on the data within ballast levels. A library program available at the University of Alberta Computing Centre was used (Grobben, 1970).

The model was:

$$Y = a + b_1 s + b_2 s^2 + b_3 s^3 + b_4 g + b_4 g^2$$ where Y = the performance parameter, s = slip, and g = rear wheel speed

The exact form of relationships between the parameters and slip was not known. Heady and Diltron (1961) state that if the algebraic form of a function is unknown, a polynomial will give an approximation quite similar to the true function over the range of interest. The degree of polynomial to be fitted is of importance. Heady and Diltron indicate that within a region occupied by the data, a first or second order polynomial offers an adequate fit. Sometimes it is worthwhile to estimate third degree terms if more precision is required and the data covers a wider range.

6.4 Experimental Error Analysis

An analysis of the experimental error in a representative set of data for the tests indicate the maximum error to be generally less than 2%.

7. RESULTS AND DISCUSSION

7.1 Coefficient of Net Traction

The analysis of covariance for the coefficient of net traction is given in table 1.

The results showed significant differences between levels of ballast, surface conditions and gears. None of the interactions were significant.

The means of the coefficients of net traction for the various treatment levels are given in table 2. The means were adjusted to a common value of slip by the analysis of covariance to provide a basis for between and within treatment comparisons. The means in table 2 have been adjusted to 37% slip. This slip level was governed by the data. 37% was the overall mean for slip in this experiment.

Using appropriate procedures as outlined in Steel and Torrie (1960), Duncan's New Multiple Range Test, or Duncan's test, was carried out on the means. The results are shown in table 3. The numbers represent levels of increasing ballast and the means are listed in ascending order.

Levels one to six of ballast seem to have the same coefficient of net traction (5% probability level). Increasing the amount of ballast decreases the coefficient of net traction. Domier (1966) obtained similar results on Osborne clay soil.

Gears 2,3, and 4 had the same coefficients of traction. Traction performance is generally assumed to be constant over the normal field speed range (Gill and Vanden Berg, 1967). In this study the low speed had a different coefficient of traction than the speeds closer to the normal working range.

TABLE 1: ANALYSIS OF COVARIANCE FOR COEFFICIENT OF NET TRACTION (μ).

Source of Variation		Degrees of Freedom	Mean Squares	F
Ballast	B	8	0.0834	6.359**
Surface	S	1	0.0845	6.443*
	B x S	8	0.0249	1.896
Error (1)	R/B x S	36	0.0131	
Gear	G	3	0.0516	6.445**
	G x B	24	0.00641	0.802
	G x S	3	0.00223	0.279
	G x B x S	24	0.00719	0.879
Error (2)	G x R/B x S	108	0.0080	

* Significant at the 5% probability level.

** Significant at the 1% probability level.

TABLE 2: MEANS FOR THE COEFFICIENT OF NET TRACTION* (μ).

Ballast level	Grass surface	Fallow surface	Average over surfaces
1	0.480	0.423	0.452
2	0.468	0.442	0.455
3	0.466	0.420	0.443
4	0.466	0.404	0.434
5	0.469	0.422	0.446
6	0.461	0.435	0.448
7	0.435	0.388	0.412
8	0.412	0.379	0.396
9	0.426	0.333	0.379
Avg. over ballasts	0.454	0.405	0.430
Gear	**Grass surface**	**Fallow surface**	**Average over surfaces**
1	0.460	0.378	0.419
2	0.469	0.391	0.430
3	0.473	0.394	0.434
4	0.477	0.393	0.435
Avg. over gears	0.470	0.389	0.430

* The means are adjusted to 37% slip by the analysis of covariance.

TABLE 3: DUNCAN'S TEST ON THE MEANS OF THE COEFFICIENT OF NET TRACTION (μ).

a) Ballast

Grass:	8	9	7	6	4	3	2	5	1
Fallow:	9	8	7	4	3	5	1	6	2
Over both surfaces:	9	8	7	4	3	5	6	1	2

b) Gears

Grass:	1	2	3	4
Fallow:	1	2	4	3
Over both surfaces:	1	2	3	4

Means underscored by the same line are not significantly different at the 5% probability level.

The numbers represent levels of ballast and gear. The means for the coefficients of traction for the levels of gear and ballast are listed in ascending order.

A stepwise multiple regression of the data at each ballast level was made. The form of the polynomial fitted was

$$y = a + b_1S + b_2S^2 + b_3S^3 + b_4g + b_5g^2$$

as outlined in section 6.3,

where ; S = slip

and g = rear wheel speed.

Figures 11 and 12 are graphs of some coefficients of net traction. The squared multiple correlation coefficients (R^2) are shown on the graphs. A rear wheel speed of 2 mph was used.

7.2 Tractive Efficiency

The analysis of covariance for tractive efficiency is given in table 4. Significant differences in efficiency occurred between levels of surface,ballast and gears. Significant interactions existed between ballast and surface and between gears, ballast and surface.

The means for tractive efficiencies due to different treatments are given in table 5.

The significant ballast by surface interaction is caused by the magnitude difference between the efficiencies on the grass and fallow surfaces and by the fact that on the fallow surface the efficiency begins to decrease more rapidly than on the grass surface for the higher levels of ballast.

Duncan's test on the means of the tractive efficiencies is shown in table 6.

Most of the gears had different efficiencies. Gears one and two

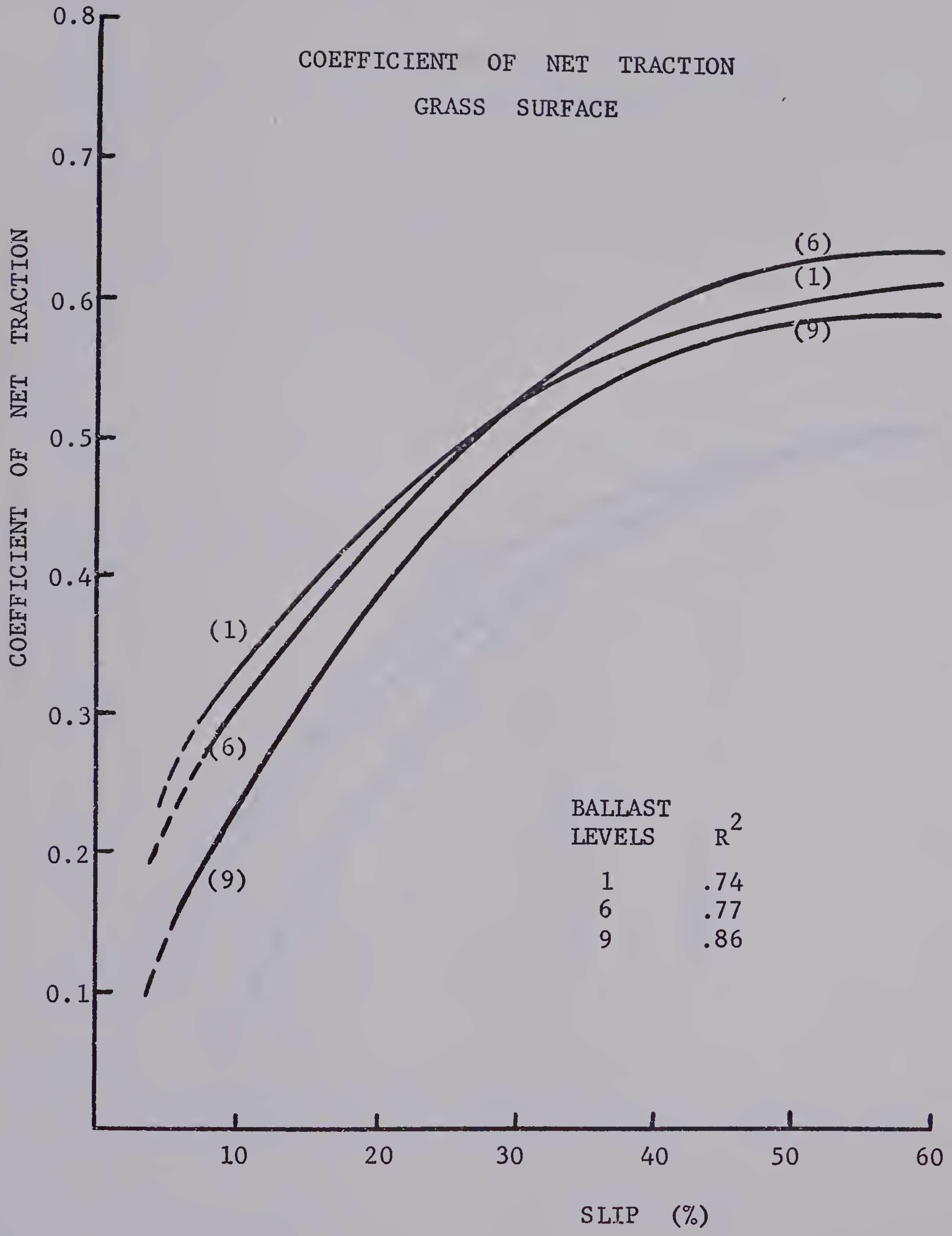

Figure 11: Coefficient of net traction on grass surface.

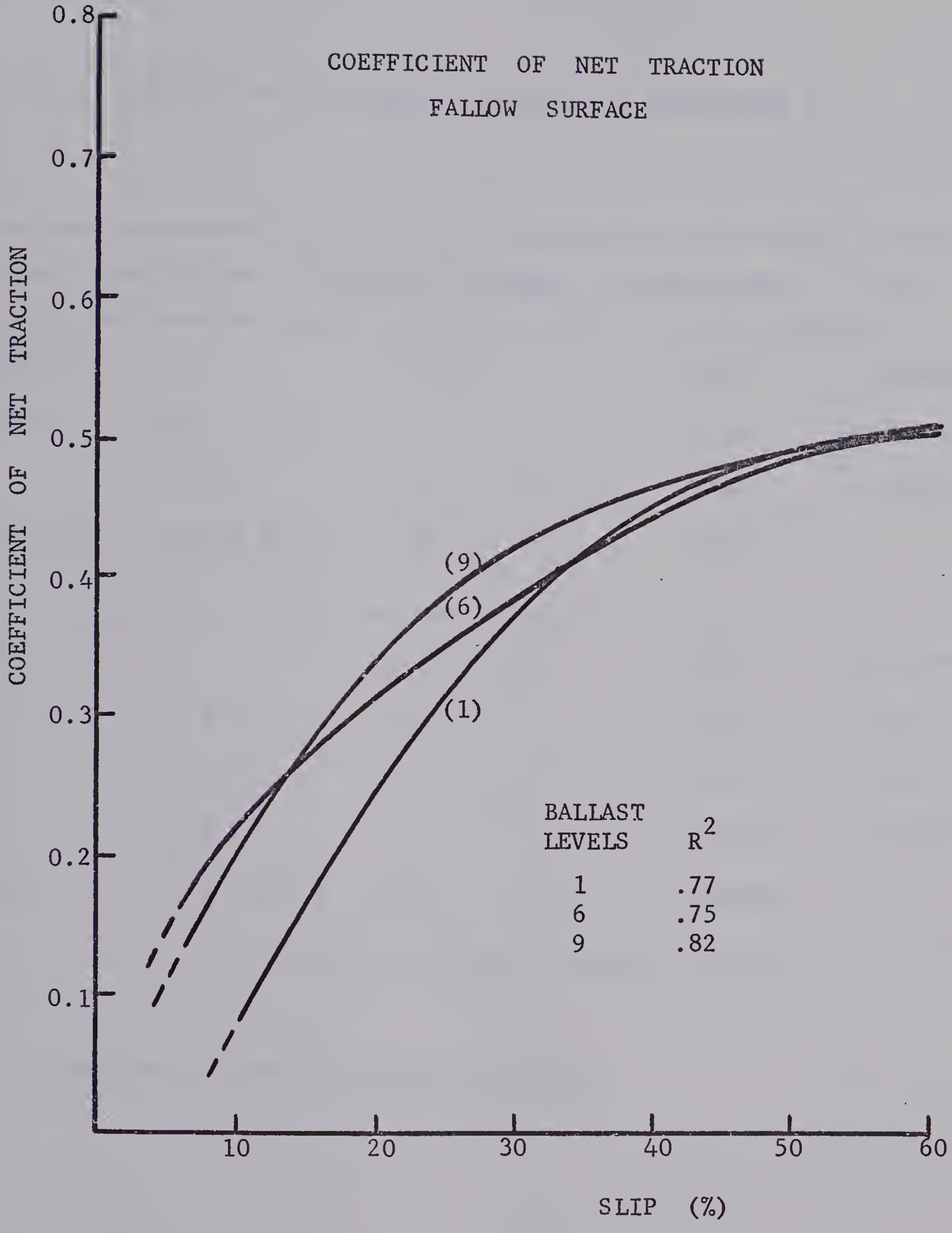

Figure 12: Coefficient of net traction on fallow surface.

TABLE 4: ANALYSIS OF COVARIANCE FOR TRACTIVE EFFICIENCY (η).

Source of variation		Degrees of freedom	Mean squares	F
Ballast	B	8	0.125	4.558**
Surface	S	1	0.418	15.211**
	B x S	8	0.998	3.630**
Error (1)	R/B x S	36	0.0275	
Gear	G	3	0.366	41.170**
	G x B	24	0.0134	1.512
	G x S	3	0.0130	1.460
	G x B x S	24	0.0344	3.874**
Error (2)	G x R/B x S	108	0.00889	

** Significant at the 1% probability level.

TABLE 5: MEANS FOR THE TRACTIVE EFFICIENCY (η).

Ballast level	Grass surface	Fallow surface	Average over surfaces
1	0.465	0.409	0.437
2	0.471	0.410	0.441
3	0.483	0.444	0.463
4	0.509	0.434	0.471
5	0.491	0.441	0.465
6	0.509	0.462	0.486
7	0.474	0.455	0.464
8	0.469	0.469	0.469
9	0.494	0.337	0.434
Avg. over ballasts	0.485	0.429	0.459

Gear	Grass surface	Fallow surface	Average over surfaces
1	0.507	0.458	0.483
2	0.482	0.449	0.466
3	0.471	0.434	0.453
4	0.455	0.417	0.436
Avg. over gears	0.479	0.440	0.459

* The means are adjusted to 37% slip by the analysis of covariance.

TABLE 6: DUNCAN'S TEST ON THE MEANS OF TRACTIVE EFFICIENCY (η).

a) Ballast

Grass:	1	8	2	7	3	5	9	4	6
Fallow:	9	1	2	4	5	3	7	6	8
Over both surfaces:	9	1	2	3	7	5	8	4	6

b) Gears

Grass:	4	3	2	1
Fallow:	4	3	2	1
Over both surfaces:	4	3	2	1

Means underscored by the same line are not significantly different at the 5% probability level.

The numbers represent levels of ballast and gear. The means for the tractive efficiencies for the levels of gear and ballast are listed in ascending order.

on fallow and gears 2 and 3 on grass had the same efficiencies. Efficiencies remained the same for wide ranges of ballast. The tractive efficiency on the fallow surface was more sensitive to ballast changes. King (1948) made similar observations for traction tests.

A stepwise multiple regression was carried out using the same model as indicated in section 7.1.

Graphical results are shown in figures 13 and 14. The squared multiple correlation coefficients (R^2) are shown. The rear wheel speed was 2 mph.

On the fallow surface the maximum efficiencies usually occurred at high slip values, approximately 20 - 30%. At the ballast level where the maximum efficiency was obtained the maximum occurred at about 10% slip.

On the grass surface the maximum efficiency occurred at a low value of slip. It is a limitation of the model and of the data (weighted towards the larger slip values) that the regression curves do not tend towards zero in the low slip range.

The analysis of covariance on tractive efficiency indicated the optimum ballast to be 6 levels on grass and 6 over both surfaces. On fallow 8 was the best followed by 6, however there was no statistical difference between 8 and 6 levels of ballast on the fallow surface.

Table 7 is an indication of how the tractive efficiency varied due to ballast changes in the normal working slip range. The regression equations were integrated to determine the average efficiencies over the 10 - 30% range of slip.

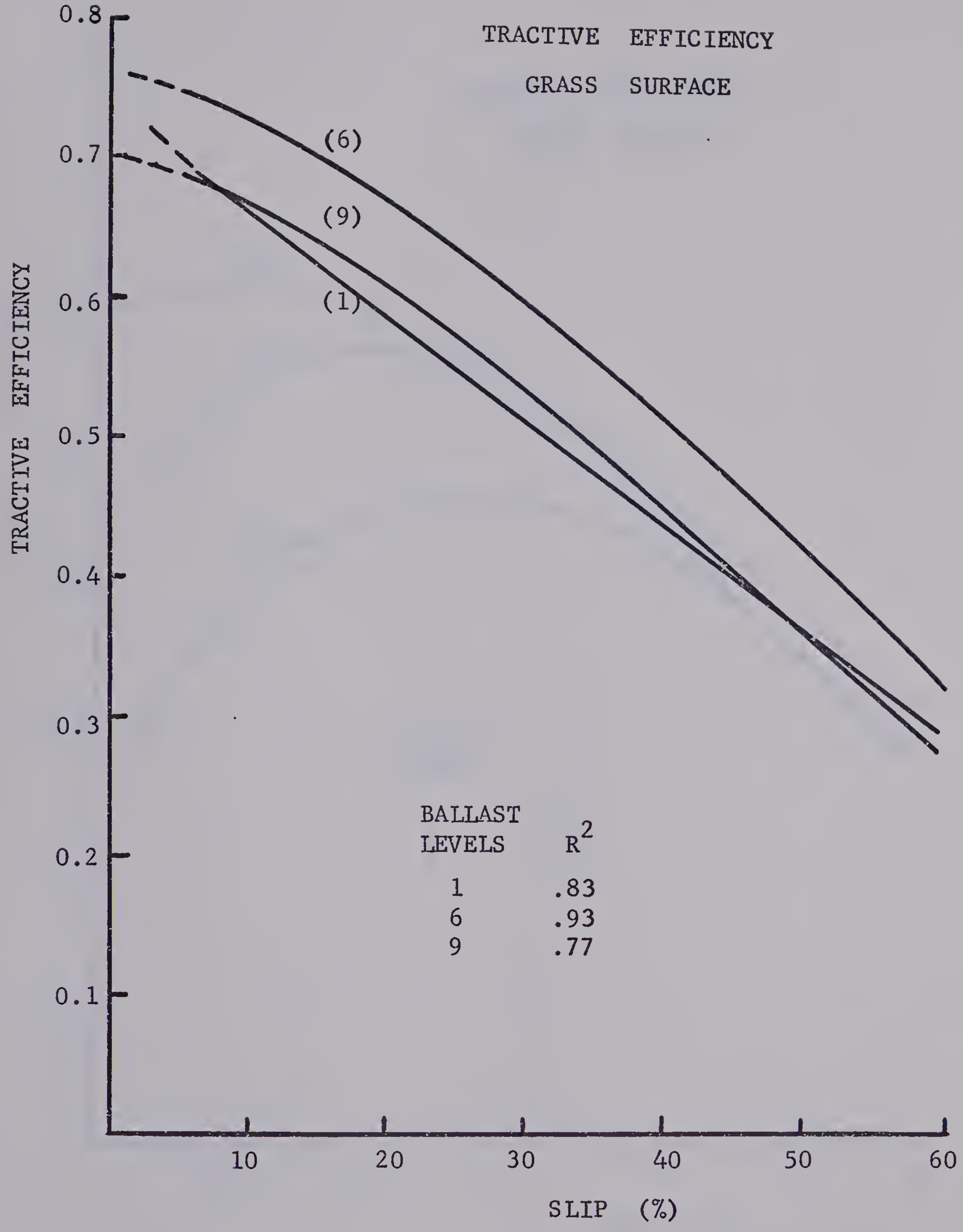

Figure 13: Tractive efficiency on grass surface.

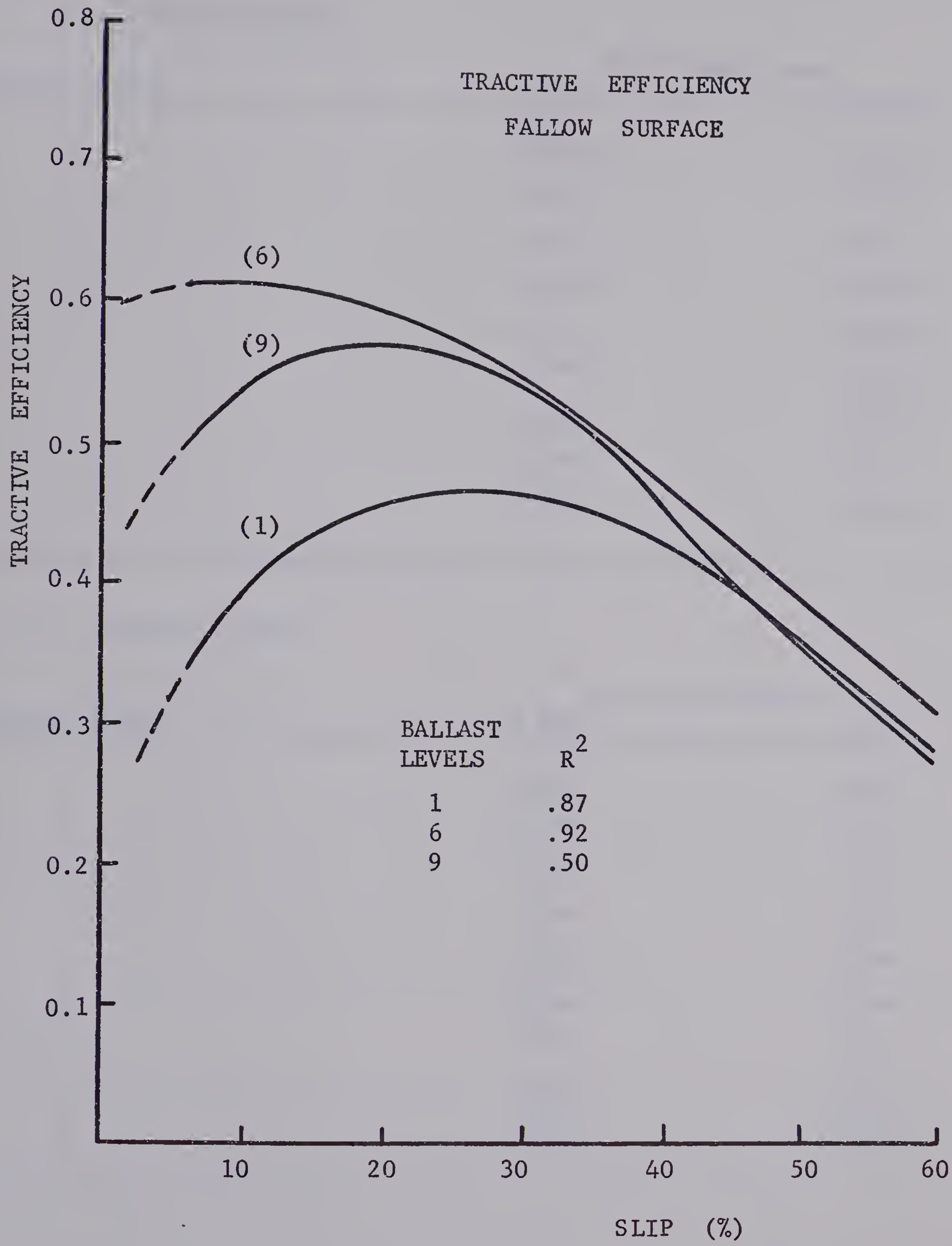

Figure 14: Tractive efficiency on fallow surface.

TABLE 7: AVERAGE TRACTIVE EFFICIENCIES FOR THE 10 - 30% SLIP RANGE.

a) Fallow Surface

Ballast level	Rear Wheel Speed 2 mph	Rear Wheel Speed 3 mph
1	0.430	0.412
2	0.415	0.404
3	0.546	0.527
4	0.493	0.481
5	0.541	0.526
6	0.580	0.553
7	0.440	0.452
8	0.446	0.468
9	0.538	0.432

b) Grass surface

Ballast level	Rear Wheel Speed 2 mph	Rear Wheel Speed 3 mph
1	0.589	0,566
2	0.609	0.554
3	0.633	0.573
4	0.649	0.603
5	0.630	0.590
6	0.668	0.565
7	0.587	0.587
8	0.604	0.575
9	0.606	0.672

On the fallow surface the best efficiency occurred at 6 levels of ballast. This point is the level before tire pressure was changed due to allowable load constraints of the tire (14 to 16 psi).

The grass surface showed a similar effect for a 2 mph speed but at 3 mph the regression indicated that the higher ballasts resulted in greater efficiencies.

The pressure change may be an important factor. Freitag (1965) found that, except for large tire loads, lower inflation pressures resulted in improved performance on non-frictional soils. Kliefoth (1966) noted that the traction coefficient decreased at higher inflation pressures. McLeod et al (1966) found that lower inflation pressure made little difference with respect to pull but the tractive efficiency was improved.

7.3 Drawbar Horsepower

The analysis of covariance for drawbar horsepower is shown in table 8. Significant differences exist between the main effects of ballast, surface and gears. The gear by ballast and the gear by surface interactions were significant.

The means for drawbar horsepower due to the treatments are given in table 9.

The significant gear by ballast interaction is caused from the fact that the horsepower at 8 levels of ballast (fallow) is less than that at 7 levels of ballast for some gears, however Duncan's test (table 10) shows no significant differences between drawbar horsepower for ballast levels of seven and eight.

TABLE 8: ANALYSIS OF COVARIANCE FOR DRAWBAR HORSEPOWER (DBHP).

Source of variation		Degrees of freedom	Mean squares	F
Ballast	B	8	55.7	10.23**
Surface	S	1	3.21	58.85**
	B x S	8	3.76	0.690
Error (1)	R/B x S	36	5.45	
Gear	G	3	4838	1027.9**
	G x B	24	15.4	3.267**
	G x S	3	127	26.989**
	G x B x S	24	4.28	0.910
Error (2)	G x R/B x S	108	4.706	

** Significant at the 1% probability level.

TABLE 9: MEANS FOR THE DRAWBAR HORSEPOWER* (DBHP).

Ballast level	Grass surface	Fallow surface	Average over surfaces
1	7.17	5.41	6.29
2	7.39	6.20	6.80
3	7.78	6.37	7.08
4	8.51	6.88	7.69
5	9.00	7.49	8.25
6	9.42	8.25	8.84
7	10.00	8.86	9.43
8	10.09	8.66	9.38
9	10.72	9.04	9.88
Avg. over ballasts	8.90	7.46	8.18

Gear	Grass surface	Fallow surface	Average over surfaces
1	6.00	5.30	5.65
2	7.92	6.96	7.44
3	9.16	7.91	8.54
4	12.13	10.05	11.09
Avg. over gears	8.80	7.56	8.18

* The means are adjusted to 37% slip by the analysis of covariance.

TABLE 10: DUNCAN'S TEST ON THE MEANS OF THE DRAWBAR HORSEPOWER (DBHP).

a) Ballast

Grass:	1	2	3	4	5	6	7	8	9
Fallow:	1	2	3	4	5	6	8	7	9
Over both surfaces:	1	2	3	4	5	6	8	7	9

b) Gears

Grass:	1	2	3	4
Fallow	1	2	3	4
Over both surfaces:	1	2	3	4

Means underscored by the same line are not significantly different at the 5% probability level.

The numbers represent levels of ballast and gear. The means for the drawbar horsepowers are listed in ascending order.

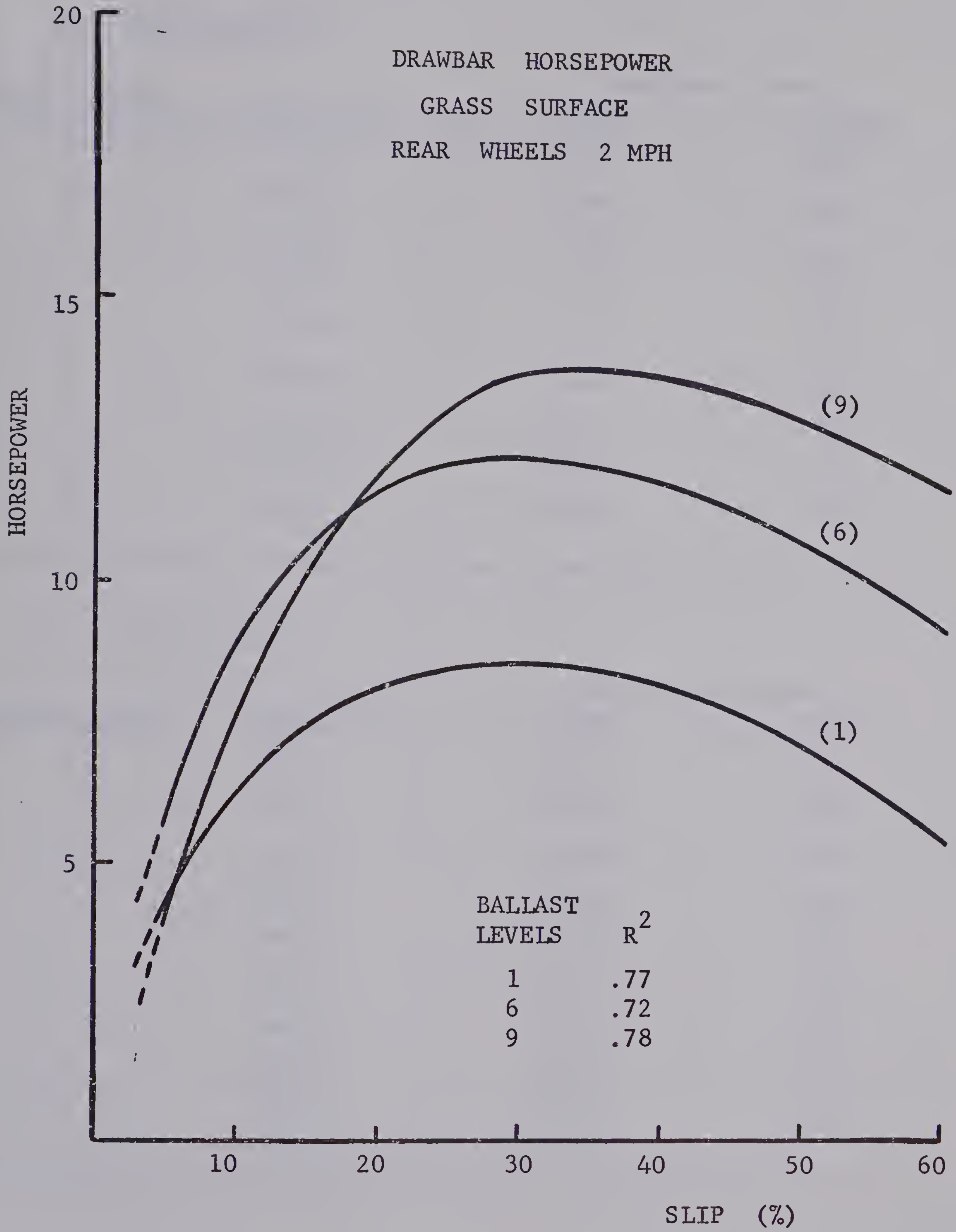

Figure 15: Drawbar horsepower on grass surface.

TABLE 11: MAXIMUM DRAWBAR HORSEPOWERS.

a) Grass Surface

Ballast level	Slip (%)	Rear Wheel Speed 2 mph	3 mph
1	29.9	8.32	12.1
2	30.5	9.26	12.3
3	30.6	9.85	14.0
4	30.9	11.0	14.5
5	33.2	11.1	15.6
6	33.5	12.3	16.0
7	37.6	12.4	17.4
8	34.6	12.7	17.1
9	34.8	13.9	17.9

b) Fallow Surface

Ballast level	Slip (%)	Rear Wheel Speed 2 mph	3 mph
1	39.3	6.25	8.51
2	42.5	6.25	8.88
3	38.3	7.29	10.4
4	40.7	7.26	10.8
5	41.4	8.38	12.4
6	34.5	9.07	12.8
7	40.4	9.89	15.4
8	41.7	9.76	15.3
9	31.7	12.0	17.4

The significant gear by surface interaction is caused by the magnitude difference in drawbar horsepower between surfaces.

The results of Duncan's test for the drawbar horsepower are given in table 10.

A stepwise multiple regression using the model of section 7.1 was run. Graphical results are shown in figure 15 for the grass surface to indicate the trends found. The squared multiple correlation coefficients are shown.

An analysis of the regression equations was carried out to determine the maximum drawbar horsepowers and the slips at which these occurred. The results are shown in table 11.

The drawbar horsepower was still increasing at nine levels of ballast indicating that for the gears tested soil characteristics and not engine horsepower was the limiting factor.

7.4. Tractive Power Coefficient.

The analysis of covariance for the tractive power coefficient is given in table 12. Significant differences existed between levels of ballast, surface and gears. The interaction between gear and surface was significant. This was due to a magnitude difference between surfaces and due to the fact that on the grass surface the tractive power coefficient increased through the gears while on the fallow surface it decreased for the fourth gear.

The means for the tractive power coefficient are given in table 13.

The results of Duncan's test on the means are given in table 14. The results indicate that the highest π value should occur at the lower gears. On the fallow surface the ballast level at which the

TABLE 12: ANALYSIS OF COVARIANCE FOR TRACTIVE POWER COEFFICIENT (π).

Source of variation		Degrees of freedom	Mean squares	F
Ballast	B	8	0.0514	8.19**
Surface	S	1	0.395	62.86**
	B x S	8	0.00972	1.548
Error (1)	R/B x S	36	0.00628	
Gear	G	3	0.0254	6.127**
	G x B	24	0.00409	0.993
	G x S	3	0.182	44.10**
	G x B x S	24	0.00465	1.13**
Error (2)	G x R/B x S	108	0.00412	

** Significant at the 1% probability level.

TABLE 13: MEANS FOR THE TRACTIVE POWER COEFFICIENT* (π).

Ballast level	Grass surface	Fallow surface	Average over surfaces
1	0.300	0.235	0.266
2	0.280	0.232	0.256
3	0.273	0.230	0.251
4	0.279	0.220	0.249
5	0.273	0.227	0.250
6	0.268	0.234	0.251
7	0.257	0.215	0.236
8	0.248	0.206	0.227
9	0.259	0.200	0.229
Avg. over ballasts	0.271	0.222	0.246
Gear	**Grass surface**	**Fallow surface**	**Average over surfaces**
1	0.263	0.215	0.239
2	0.271	0.223	0.247
3	0.275	0.223	0.249
4	0.279	0.221	0.250
Avg. over gears	0.272	0.221	0.246

* The means have been adjusted to 37% slip by the analysis of covariance.

TABLE 14: DUNCAN'S TEST ON THE MEANS OF THE TRACTIVE POWER COEFFICIENT (π).

a) Ballast

Grass:	8	7	9	6	5	3	4	2	1
Fallow:	9	8	7	4	3	2	6	1	
Over both surfaces:	8	9	7	4	5	6	3	2	1

b) Gears

Grass:	1	2	3	4
Fallow:	1	2	3	4
Over both surfaces:	1	2	3	4

Means underscored by the same line are not significantly different at the 5% probability level.

The numbers represent levels of ballast and gear. The means for the tractive power coefficients are listed in ascending order.

highest π occurred is also the level at which the best efficiency occurred.

Gill and Vanden Berg (1967) and Persson (1967) state that the tractive power coefficient indicates how much output power a wheel produces with a given dynamic weight. The pull at which the maximum π occurs should be the pull at which the maximum work is delivered. It should be noted that the maximum π is not the maximum power output that the wheel produces with respect to levels of ballast. The tractor produced high drawbar horsepowers at high levels of ballast and high π values at the lower ballast levels. From an examination of the data, however it seems that the maximum π values within each ballast level occurred at nearly the same slip values that the maximum drawbar horsepower was developed (30 - 35% on grass and 35 - 40% on fallow).

7.5 Optimum Ballast

The best ballast with respect to maximum pull and maximum drawbar horsepower seemed to be the maximum load carrying capacity of the tire. Within the range of gears and ballasts studied the engine did not limit the output power.

In terms of a load factor discussed by Domier (1966), the best tractive efficiencies occurred at a 100% load factor of the tire. The load factor is the ratio of tire load to the load recommended as a maximum by tire manufacturers for a given tire pressure. This may indicate that at a constant pressure (for this soil) the maximum tire load recommended gives the best efficiency. In this study however, there was little change (statistically) in efficiency over a wide ballast range.

In the range of ballast where the best efficiency occurred, ballast

had little effect on the coefficient of net traction. Therefore it seems reasonable to use coefficients of traction and tractive efficiency values predicted by the regressions to determine a ballast level required for maximum output performance.

The problem is to determine the best slip for a particular field operation. Historically 16% has been recommended as a maximum due to wear restrictions but values in a range of 10 - 20% have been noted by Williams and Van Syoc (1968).

This study indicated the highest work outputs occurring near 30% slip while best efficiencies may occur at less than 10% slip (grass surface). The actual values of slip at which the best tractive efficiency occurred could not be determined due to limitations of the regression model. Operating at a high work output will require high slip values (tire wear may be a constraint) and will result in lower tractive efficiencies.

A model is proposed based on the performance at the level of ballast where the best tractive efficiency occurred. Efficiencies and the coefficients of traction are given for 15% slip and 20% slip. The model will predict the ballast required to utilize all of the engine power.

The output performance of a tractor is

$$\text{DBHP} = \text{tractive efficiency } (\eta) \times \text{Axle Horsepower.}$$

Zoz (1972) gives typical transmission efficiencies in which PTO horsepower x 0.95 = Axle Horsepower for gear type transmissions. Field speed = (No slip speed) x (1 - Slip) where slip is expressed as a decimal.

$$1 - \text{Slip} = 0.85 \text{ for } 15\% \text{ slip}$$

$$1 - \text{Slip} = 0.80 \text{ for } 20\% \text{ slip.}$$

$$\text{Dynamic rear weight} = \frac{(\text{PTO horsepower}) \times 0.95 \times \eta \times 375}{\mu \times (\text{No Slip speed}) \times (1 - \text{Slip})}$$

Static rear weight = Dynamic rear weight - Weight transfer.

The expression used for weight transfer is:

$$\text{Weight transfer} = \text{Net pull} \times \frac{\text{Drawbar height}}{\text{wheel base}}$$

for a horizontal pull. Pull is expressed in pounds, drawbar height and wheel base in feet. Net pull = μ x Dynamic weight.

This form of expression is not strictly correct as was demonstrated by Townsend et al (1971) in that weight transfer is underestimated. A more accurate estimate depends on a knowledge of torque input to the tractor wheels.

The expression for rear static weight is then:

$$\text{Rear Static Weight} = (1 - \mu \frac{H}{W}) \times \frac{\text{PTO} \times 0.95 \times \eta \times 375}{\mu \times G \times (1 - S)}$$

where μ = coefficient of net traction,

η = tractive efficiency,

H = drawbar height (feet),

W = wheel base (feet),

PTO = power takeoff horsepower,

G = no slip speed (miles per hour),

1 -S = .85 for 15% slip and .80 for 20% slip.

The traction properties are given in table 15. The regression equations are only valid near the speed range of 1.5 - 3 mph. The coefficients at 3 mph are given in table 16. The coefficient of traction from the analysis of covariance seemed to be nearly independent of speed at higher speeds. Thus at speeds greater than 3 mph it may be necessary to compromise and use the values of the coefficient at 3 mph. This

TABLE 15: TRACTION PROPERTIES**

a) Grass Surface

Tractive efficiency (η) = $A - 0.009G - 0.019G^2$

where $A = 0.797$ for 15% slip,

and $A = 0.766$ for 20% slip.

Coefficient of Net Traction (μ) = $B + 0.284G - 0.066G^2$

where $B = 0.087$ for 15% slip,

and $B = 0.141$ for 20% slip.

b) Fallow Surface

Tractive efficiency (η) = $A - 0.169G + 0.029G^2$

where $A = 0.830$ for 15% slip,

and $A = 0.815$ for 20% slip.

Coefficient of Net Traction (μ) = $B - 0.058G + 0.013G^2$

where $B = 0.347$ for 15% slip,

and $B = 0.387$ for 20% slip.

* G refers to the no slip speed.

** The properties are based on regression equations and to be strictly valid the no slip speed should be near the range 1.5 - 3 mph.

TABLE 16: TRACTION PROPERTIES AT 3 MPH.

a) Grass

15% slip	20% slip
$\eta = 0.60$	$\eta = 0.57$
$\mu = 0.34$	$\mu = 0.40$

b) Fallow

15% slip	20% slip
$\eta = 0.58$	$\eta = 0.56$
$\mu = 0.29$	$\mu = 0.33$

reasoning is not as valid when considering the tractive efficiency which was speed dependent.

For the tractor used in this study the prediction equation indicates that tire constraints prevented a high enough ballast level to allow all the engine power to be used at low speeds.

7.6 Limitations of Equipment and Test Techniques

Slip is a difficult variable to measure. Vanden Berg and Reed (1961) discuss the inherent difficulties involved in the measurement of slip and the determination of a zero slip definition. The authors cite Phillips (1961) who suggests a zero slip measurement may be based on:

1. the distance travelled by one revolution in the self-propelled state;
2. the distance travelled by one revolution in the zero torque state and;
3. the distance travelled when the instantaneous center of rotation of the wheel is at the undisturbed surface of the medium on which the wheel is operating.

The difficulties regarding the definition arise from the fact that zero pull and zero torque never occur simultaneously, rolling radius changes under load, sinkage has an effect and a slip differential may exist across a section of the tire. Slip measurement at best is an estimate based on an observation of external velocities of the device.

In this study error exists in slip measurement possibly due to soil variation and load effects. No estimate was possible on the latter factor which tends to change the effective rolling radius. Error due to soil

variation was not estimated, however slip calibrations (section 4.3) showed good linear correlations between forward velocity and voltage produced by the electric generators. Ten to fifteen 100 foot runs were made for each calibration.

The load car had a high rolling resistance relative to the tractive ability of the tractor. The maintenance of low slip conditions at the beginning of each run was difficult. The worst case was at a low level of ballast on the loose surface. This had an influence on the results of the regression analysis as the data tended to be weighted towards higher values of slip.

The load car placed a restriction on speed due to some instability at high speeds in terms of the wheels not following a perfectly straight path. The surface condition may have had an effect. The analysis of data for higher speeds had too much variation to merit complete analysis.

The load car had two important advantages. First it was quiet, in comparison to the use of a tractor as a loading unit which allowed the operator of the load car to hear the test tractor engine. Secondly, it was easy to obtain a smooth, gradual increase in pull over a long distance.

8. SUMMARY

Surface, ballast, and speed all had significant effects on the performance parameters (μ, η, π, DBHP).

On the grass surface the coefficients of traction (μ) for ballast levels 1 - 6 (3470 - 5290 lbs rear weight) were not significantly different. On the fallow surface the same observation was made with the exception of ballast level 4 which had a significantly lower coefficient of traction. The higher ballast levels (7,8,9) which required higher tire pressures, had significantly lower coefficients of traction. The tire specifications indicate that for a 4 ply tire, 6 levels of ballast is the maximum load. There was no significant difference in the coefficient of traction between gears 2,3, and 4. Gear 1 had a significantly lower coefficient. The data seems to indicate that speed may have an effect on the coefficient of traction. The grass surface had a higher coefficient than the fallow surface.

The tractive efficiency (η) was highest at low speeds. The efficiencies were statistically different with respect to gear with the exception of gears 2 and 3 which had the same efficiency on grass and gears 1 and 2 which had the same efficiency on fallow. The trends with respect to ballast and efficiency are not clear, however if one considers levels 1 - 6 of ballast (14 psi), the efficiency increased slightly with increasing weight. The change in tire pressure to 16 psi at 7 levels of ballast caused a decrease in efficiency. This study could not determine the slip value for which the maximum efficiency was found on the grass surface.

The tractive power coefficient (π) seems to be useful not by its magnitude but by the point of operation (slip and pull) at which π is a maximum. The data in this study indicated that the

maximum π occurred at nearly the same slip values as the maximum drawbar horsepower occurred.

The drawbar horsepower (DBHP) and pull increased with increasing ballast for the gears used. Limitations of the test equipment prevented the use of gears above 3 mph. This study did not reach a point where high ballast levels along with high speeds would cause decreasing drawbar horsepowers due to high rolling resistances and engine power limitations.

The expression proposed for optimum ballast is the ballast required to utilize all the engine power. At low speeds this may not be possible. in the case where the tire cannot carry the calculated load or may not be desirable from the point of view of the mechanical strength of the tractor. The expression is compromise between efficiency and work output. The operating points are 15 and 20% slip. On grass, higher tractive efficiencies were found below 15% slip but higher work outputs occurred above this point.

With respect to efficiency the optimum ballast was 2750 pounds (6 levels) above the tractors unballasted weight (14 psi tire pressure). With respect to power output the maximum power delivered (under 3 mph) in these tests occurred at the maximum load carrying capacity of the tire (9 levels).

The statistical tests proved to be useful, however their limitations should be recognized. The main difficulty in comparing traction performance curves is the decision whether to compare performance at one point or to compare the entire curves. The latter point is more difficult and is partly overcome by the analysis of covariance which carries out a regression of the data.

9. SUGGESTIONS FOR FURTHER WORK

Careful attention should be given to the area close to the origin when studying the curves of the various parameters with respect to slip. A method of slip measurement which can accurately determine slip in the 0 - 16% range is required if one is to study the behaviour of parameters such as tractive efficiency near zero slip.

The problem of what happens with respect to efficiency when the tractor is ballasted for maximum output and is used for light work requires closer attention.

A problem exists concerning the size of the test tractor. A quantitative description of the size effect may be useful for prediction of the tractive performance of larger tractors.

The influence of other soil types found in Alberta may be of interest concerning their effect on the performance parameters.

If one uses an analysis of covariance technique and is interested in a particular level of slip, care should be taken so that the overall mean of slip is the same as the level of interest. This is necessary if a polynomial, such as was used in this study, is used in a regression of the data.

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11. APPENDICES

APPENDIX 1: EQUIPMENT SPECIFICATIONS

1. Tractor

Model: Massey-Ferguson MF-135

Serial Number: 9A JI5375

Unballasted weight: front 1530 lbs.

rear 2530 lbs.

Wheel base: 72 3/8 in.

Front tread: 60 in. (for this study)

Rear tread: 56 in. (for this study)

Engine: Perkins Diesel

3 cylinder, 152.7 cu. in.,

3 plow, 38.5 pto h.p. at 2000 rpm

115 ft-lbs. torque

Tires: front 6.00-16 4 ply rating

rear 14.9-24 4 ply rating

Advertised speeds (13.6 - 28 tires) at 2000 rpm:

Gears 1,2,3,4: 1.38, 1.80, 2.07, 2.70 (mph)

2. Pull Transducer

Circular O-Ring pull transducer

Dimensions: diameter 4 in.

cross-section 9/16 in x 9/16 in.

3. Strain Gauges

type EA-06-250BG-120

(typical specifications 120.0$\pm$0.15% ohms

gauge factor 2.095$\pm$0.5% at 75°F)

4. DC Generators

ripple 3% rms

output 10 v per 1000 rpm

5. Accudata 104 DC amplifier (Honeywell)

gain: 0 - 250 continuously variable

DC gain accuracy: better than $\pm$ 0.5%

linearity: better than $\pm$0.1% of full scale output

drift: less than $\pm$ 10 μv.

6. Accudata 105 Gage control unit (Honeywell)

excitation: 3.5 - 11.5 v for 350 Ω

1.5 - 5 v for 120 Ω

Compensates for $\pm$ 5% unbalance

drift: $\pm$ 0.05% in 24 hrs.

7. Ultra violet recorder type SE 2005 (SE Laboratories, England)

direct writing oscillograph

galvanometers cover frequency ranges 0 - 30 c/s

to 0 - 8000 c/s.

linearities of $\pm$ 1% over $\pm$ 6 cm deflection can be achieved.

Speeds 1.25 - 2000 mm/sec.

APPENDIX 2. TIRE SPECIFICATIONS*

14.9 - 24 rear tractor tire

Load (lbs)	Pressure (psi)
2700[4]	14
2920	16
3130	18
3330[6]	20

The small index numbers denote ply rating for which accompanying load and inflation is a maximum.

Ballast level	Static rear weight (lbs)	Tire pressure (psi)
1	3470	14
2	3830	14
3	4200	14
4	4560	14
5	4930	14
6	5290	14
7	5650	16
8	6020	18
9	6380	20

* (Goodyear Tire Company, 1970)

APPENDIX 3. SOIL PROPERTIES (top 6 inches)

1. Grass Surface

Property	Level	Standard Deviation	Units
Dry Density	54.95	8.04	lb/ft^3
Bulk Density	68.84	8.82	lb/ft^3
Moisture	25.61	4.71	%
Moisture*	28.86	4.33	%
Cone Index**	80.44	14.32	lb/in^2
C***	0.64	0.07	lb/in^2
Ø	33.39	2.37	degrees

2. Fallow Surface

Property	Level	Standard Deviation	Units
Dry Density	42.83	5.23	lb/ft^3
Bulk Density	53.91	5.66	lb/ft^3
Moisture	26.28	6.70	%
Moisture*	26.72	4.63	%
Cone Index**	60.87	17.12	lb/in^2
C***	0.35	-	lb/in^2
Ø	34.5	-	degrees

The dry density, bulk density, and moisture readings were obtained with a nuclear moisture-density surface gauge manufactured by Nuclear-Chicago.

* This is the moisture determined by a standard gravimetric procedure.

** This is the penetration resistance obtained with a 30° circular soil cone penetrometer (hand operated).

*** The C and Ø values for Coulomb's soil strength equation were obtained with a Cohron Sheargraph manufactured by Soiltest Inc., Evanston, Illinois.

The soil texture was Malmo Silty Clay.